

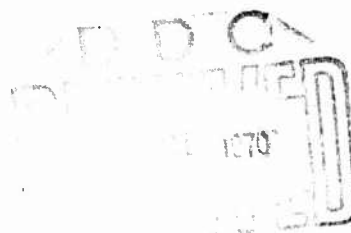
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TECHNICAL REPORT
71-5-CE

FLOW OF HEAT AND WATER VAPOR THROUGH PROTECTIVE CLOTHING

by
Ferdinand Votta, Jr.
and
Leo A. Spano



August 1970

UNITED STATES ARMY
NATICK LABORATORIES
Natick, Massachusetts 01760



CLOTHING & PERSONAL LIFE SUPPORT EQUIPMENT LABORATORY
C&PLSEL-78

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Project reference:
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FOREWORD

This preliminary report describes the design, fabrication and testing of special equipment specifically designed for investigating the transfer of heat and water vapor through protective clothing systems. This work was done by the Advanced Projects Branch, Clothing and Equipment Systems Division, Clothing and Personal Life Support Equipment Laboratory under Project No. 1J062110A533, Research on the Engineering of Combat Clothing and Equipment.

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ABSTRACT

In a continuing effort to improve the comfort and working efficiency of persons wearing various protective clothing systems, a sweating cylinder to test the heat and water vapor transfer characteristics of such systems was designed, constructed and successfully tested.

With this equipment, the heat and water vapor transfer through a carbon-treated polyurethane-nylon tricot laminated clothing system was determined under a wide range of simulated environmental conditions. Reasonably high cooling rates were obtained. With a simulated skin temperature of 91°F and air at 85°F and 50 percent R.H. flowing over the surface at 3.6 miles per hour, a cooling rate of 129 Btu per hour per square foot was obtained.

FLOW OF HEAT AND WATER VAPOR THROUGH PROTECTIVE CLOTHING

1. Introduction

One of the major problems encountered in the use of protective clothing is the thermal control of the man wearing the clothing. A man may be subject to severe over-heating if his clothing is impervious to water vapor, especially if he is performing heavy physical work. Depending on the rate of physical activity, his heat output may vary from a few hundred to over 3000 Btu/hr. Under normal conditions the body control mechanism maintains the body temperature within acceptable limits. Heat is lost from the body to the surroundings by convection to the air, by radiation, and by evaporation of sweat from the skin surface. Except in very low ambient temperatures, high rates of heat removal require the sweating mechanism. However, sweating can only produce a cooling effect when the water can evaporate from the skin surface. When the water vapor in the air next to the skin surface cannot diffuse through the protective clothing, the rate of evaporation, and thus the rate of heat removal from the skin surface, is greatly reduced.

The purpose of this investigation was to design and operate equipment for studying the rate of heat flow through various clothing systems under simulated working and environmental conditions. The equipment designed allows clothing material to be tested under a wide range of conditions. The clothing was mounted over a simulated skin surface, the temperature of which could be varied and controlled. The environmental conditions in contact with the outer surface of the clothing would also be varied and controlled. Artificial sweating was produced by a wet wick in contact with the skin surface.

So far, only one protective clothing material has been tested. This is a carbon-treated, polyurethane-nylon cloth (polyurethane foam/nylon tricot laminate, impregnated with activated charcoal sprayed on the nylon tricot side with fluoro-chemical, 7.5 oz./sq. yd.).

2. Experimental Equipment

A schematic diagram of the equipment used is shown in Figure 1. The clothing systems were tested in a chamber which consisted of the annular space formed by a brass inner pipe and an outer sleeve. The inner brass pipe was a standard 9-inch pipe welded to a disc or head at one end and a mounting flange at the other end. An asbestos cement or transite pipe was mounted over the inner pipe. The temperature drop through the transite pipe measured with a differential thermocouple, was used as the basis for calculating the rate of heat flow. The transite pipe was divided into three sections to reduce longitudinal heat transfer. The center section was one foot long and served as the test section. The total area of the transite was 3.2 square feet. Eleven thermocouples were attached to the

surfaces of the transite, each thermocouple consisting of three junctions connected in parallel. Heat was supplied by an electrically heated hot-water bath inside the inner pipe. The temperature of the bath was controlled by a potentiometric thermoregulator.

A layer of cotton shirting was mounted over the transite surface and a rubber impermeable membrane was cemented to a projection on the top flange and to the lower portion of the inner pipe. The membrane served as the simulated skin surface and also shielded the transite so that it was not wetted by the water-laden wick. The clothing system to be tested was mounted over the membrane.

The temperature and humidity of the air used in the test chamber were controlled. Room air was compressed by a 25 CFM blower. The hot air leaving the blower was cooled in a 4.3-square-foot shell and tube heat exchanger and its flow rate was measured by a rotameter. The air was then heated to the desired temperature by an electrically heated air heater consisting of a 25-inch section of one-inch tubing wound externally with a 500-watt heating tape and covered by insulation.

The humidity of the air was varied by blowing steam, produced by an electrically heated boiler, into the air in a mixing section. The air was then fed to the bottom of the outer sleeve of the test chamber and contacted the outside of the clothing. Both the air and boiler heater currents were controlled by variable transformers. The temperature and humidity of the air could be varied over a fairly wide range. However, the minimum air humidity was limited by the humidity of the room air or by the temperature of the air leaving the cooler. The minimum temperature to which the air could be cooled was approximately 64°F. Thus, except for days when the atmospheric humidity was high, air with a dew point of 64°F was the least humid air which could be produced. When the room air humidity was high, condensation took place in the cooler. This was drawn from the system's condensate drain.

The velocity of the air flowing over the surface of the clothing was varied by varying the size of the outer sleeve. In the tests to date, four different sleeves were used. The first was a 22-inch square, cross-sectional box constructed of hardboard. The others were 12-, 11- and 10-inch diameter cylinders made of galvanized iron. Each sleeve was mounted on a plywood base which was attached to the piping. The air velocities obtained in the test chamber using the different sleeves were approximately 0.092 (22"), 0.82 (12"), 1.38 (11"), and 3.6 (10") miles per hour.

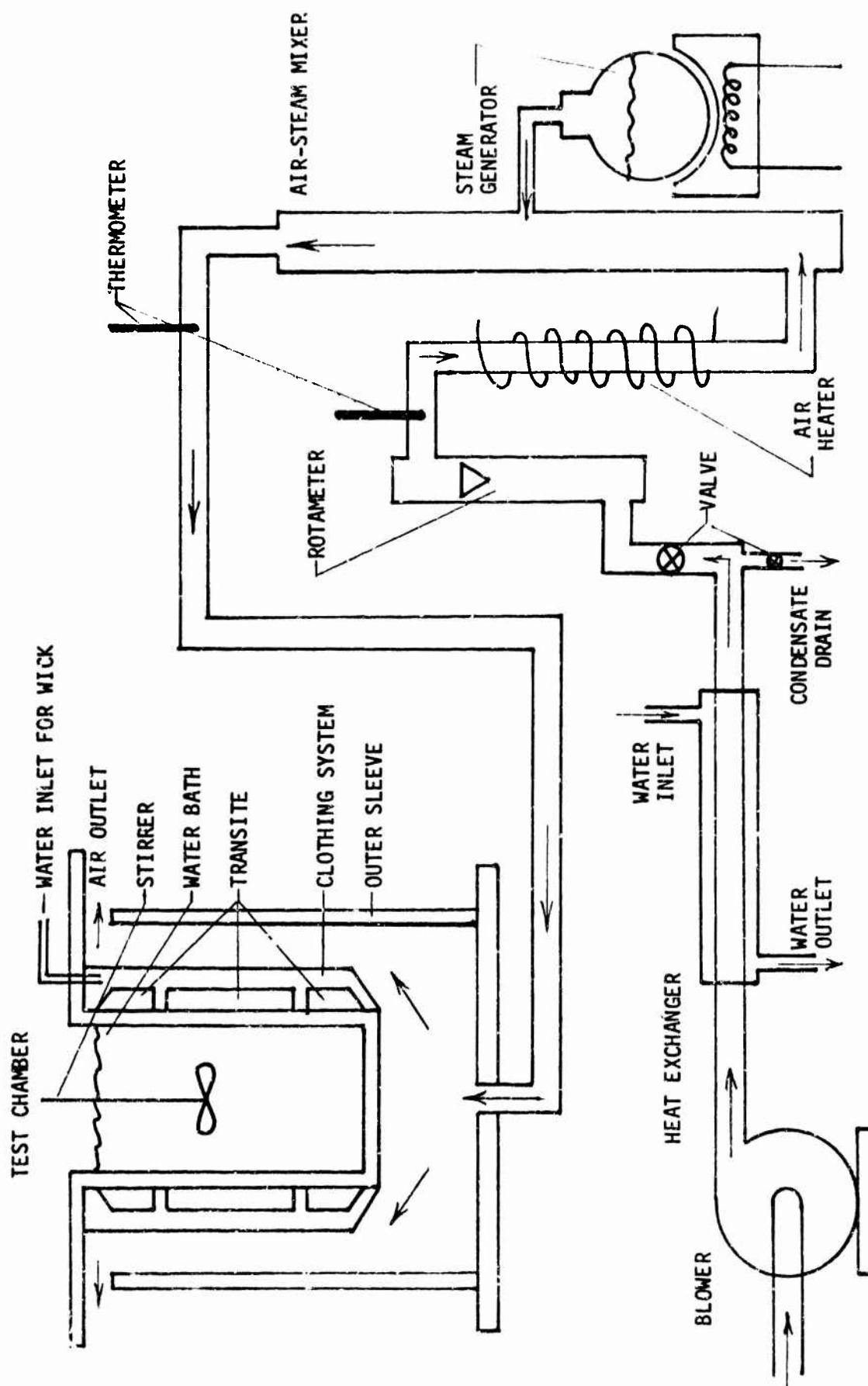


Figure 1. Diagram of Equipment

Thermocouples were installed at various points in the system. In addition to the transite and water bath thermocouples, three thermocouples were attached to the membrane (simulated skin surface), two to the wick and three to the outer surface of the clothing. Three thermocouples measured the air temperature; one of these was in the air line leaving the mixer, one in the air adjacent to the middle of the test section, and the third in the air leaving the test chamber.

Two potentiometers were used to measure temperatures. One, a 24-point Honeywell electric potential temperature recorder, was used to record the electromotive force from all thermocouple junctions in the system. The other potentiometer was a 2- pen Mosely strip chart recorder. Generally, one pen was used to measure the temperature of the membrane (simulated skin temperature) and the other to measure the temperature drop through the asbestos cement, the rate of heat flow was calculated from this temperature drop. However, by using a thermocouple selector switch, it was possible to use the more sensitive strip chart recorder to measure and record other key temperatures. Full-scale range on the strip chart recorder is adjustable. Generally, it was used with full scale set at 1 or 2 millivolts. At maximum sensitivity each division on the chart is less than 0.5°F and it was possible to estimate temperatures to approximately 0.1°F .

The humidity of the air in the test chamber was determined by "dew point cup" measurements. Two rubber tubing sample lines in the chamber were used to sample the air at each end of the test section. Air was sucked through the sample lines and blown over the dew point cup by means of an aspirator bulb. The dew points could easily be determined within 0.5°F .

3. Description of Two Clothing Systems Tested

Two clothing systems designated as Suit Nos. 7 and 8, have been tested to date. In Suit No. 7 a sheet of nylon parachute cloth (Rip-Stop 1.1-ounce OG-106) was placed over the rubber membrane skin simulator. The nylon was covered with a single sheet of carbon-treated, polyurethane-nylon cloth (polyurethane foam/nylon tricot laminate, impregnated with activated charcoal sprayed on the nylon tricot side with a fluoro-chemical, 7.5 oz/sq. yd.). The carbon-treated side was in contact with the nylon parachute cloth.

Sweating was simulated by means of a wet wick. In Suit No. 7 no wick was used. In Suit No. 8 a wick was installed over the membrane between the membrane and the nylon cloth. In other respects, Suits 7 and 8 were identical. The wick consisted of two layers of cotton toweling. Water to the wick was obtained by 12 feet of 1/16-inch Teflon tubing coiled over the membrane (between the wick and the membrane). The Teflon tubing was punched by a needle about every 2 inches. Water to the tubing was supplied from a 500-ml water reservoir.

4. Test Procedure

The water bath and stirrer were turned on and the bath was heated to the desired temperature. The bath heater was controlled by the potentiometric thermoregulator which was activated by the bath temperature. The set point was adjusted during the run to obtain the desired skin temperature. When the bath had nearly reached the proper temperature, the blower and the cooling water to the cooler were turned on. Water was added to the wick in those runs in which a wet wick was desired. The air heater and boiler heater were turned on and adjusted to obtain the desired air temperature and humidity. The air heater usually required readjustment during the run to maintain the desired air temperature. After conditions in the system had been constant for approximately 30 minutes, the temperatures were recorded and the run ended.

Four runs were made using Suit No. 8 in which no outer sleeve was used and the outer surface of the clothing was exposed to room air. In two of these tests, forced convection was obtained from a large fan placed under the equipment, while in the other two tests cooling was by natural convection. Since room air was used directly, there was no control over the temperature or humidity of the air in these four runs.

The cooling rate was calculated from the temperature drop through the transite pipe. The temperature difference between the inside and outside of the transite was measured using a differential thermocouple in the test or center section of the transite pipe. Since the thermal conductivity and thickness of the transite were known, the rate of heat transfer flow through the transite could be readily calculated.

$$q/A = \frac{k \Delta t}{L}$$

where

q/A = rate of heat flow/unit area
 k = thermal conductivity
 L = thickness of transite
 Δt = temperature drop

It was assumed that the rate of heat flow through the transite was also the rate of heat flow through the clothing.

Before carrying out this series of runs, a special test was made to test the reliability of the measurements. This was done by making an overall energy balance on the equipment. The total energy into the system came from the electrical hot water heater and from the water stirrer. The energy-out consisted of the heat transferred through the transite pipe and the loss from the water surface and top flange. The energy input of the water stirrer was estimated from the rate of heat rise of the water with only the stirrer on. The electrical energy to the heater was measured with a watt meter. The total heat transferred through the transite pipe was calculated from the temperature drop measurements, since the total surface area of the transite was known (3.8 square feet). Neglecting the heat loss from the top surface, the average value of the ratio of energy-out to energy-in for a separate tests was 0.80. In view of the fact that the heat loss from the top was neglected, this was considered a very satisfactory check of the method used to calculate the heat flow through the transite pipe.

5. Results of Testing Suits 7 and 8

The results of the tests are summarized in Table I.

Figure 2 shows the effect of air temperature on the cooling rate for suit No. 7, (no wick). In these tests the simulated skin temperature was approximately 97°F. Since there was no wick in Suit 7, the cooling represents sensible heat transfer only and changes in the humidity of the air (from 50 to 80% RH) seemed to have no effect on the cooling rate.

The effect of the relative humidity of the air and air velocity on cooling rate for Suit No. 8 is shown in Figure 3. The runs plotted (88-95) are those in which the air temperature was 85° and the skin temperature 91°F. As expected, the use of a wet wick to simulate sweating increased the cooling rate significantly. Also the cooling was strongly influenced by the air humidity and the air velocity, especially at the lower velocities.

TABLE 1
SUMMARY OF RESULTS

RUN NO.	SUIT NO.	WICK	AIR VELOCITY	OUTER SLEEVE	T _s (°F)	T _{wick} (°F)	T _{os} (°F)	T _{air} (°F)	DEW POINT (°F)	%RH	COOLING RATE (BTU/hr/ft)
59	7	none	0.092	22-in.	97.2	----	90.8	86.0	----	80	19
60					97.5	----	90.7	86.1	----	62	20
61					97.6	----	91.0	83.2	----	50	23
62					96.7	----	86.2	73.0	----	62	35
63					95.7	90.2	96.0	110.0	*75	33	64
64	8	wet			94.8	89.8	83.6	84.5	*67	50	69.5
65					90.6	89.8	85.0	84.9	*74	68	56
66					91.5	90.6	85.4	84.8	*81	85	44
67					89.4	87.2	82.3	84.9	*61	45	73
68					92.7	----	86.3	84.8	----	58	13.5
69	8	dry			89.8	89.7	82.9	84.5	67.5	55	65
70					89.8	87.9	83.5	84.5	71.5	62	59
71					91.5	90.7	87.2	85.0	83.5	94	41
72					93.0	92.2	92.2	96.2	83.5	74	26.5
73					85.2	84.6	85.3	86.3	83.5	96	0

TABLE 1 (Continued)

RUN NO.	SUIT NO.	WICK	AIR VELOCITY (mph)	OUTER SLEEVE	T _s (°F)	T _{wick} (°F)	T _{os} (°F)	T _{air} (°F)	DEW POINT (°F)	%RH	COOLING RATE (BTU/hr/ft)
74			**		92.7	91.4	90.7	89.2	74	56	40
75			4.10***	none	94.8	87.7	90.0	90.8	75	58	63
76			3.65***		90.6	86.5	83.1	86.9	69	55	82
77			**		93.0	89.6	87.0	84.5	70	60	47
78		wet			91.0	89.6	81.4	85.4	73	69	106
79					90.8	88.0	84.5	85.1	69	58	85
80					91.0	88.7	87.7	84.5	85	100	31
81					90.6	87.5	84.9	85.4	77	77	61
82	c	dry			91.4	90.0	87.7	85.3	----	----	6
83					91.1	86.9	81.1	85.3	64	49	111
84			3.60	10-in	91.2	87.3	82.4	85.3	71	63	100
85					91.4	88.5	80.0	85.1	60.5	42.5	129.5
86		wet			91.4	88.0	80.4	84.8	57	39	129.5
87					90.7	88.4	81.6	84.8	72	64.5	99
88					91.0	89.0	82.7	85.0	74.5	71	90
89					91.3	89.2	83.6	85.0	77	77.5	78

TABLE 1 (Continued)

RUN NO.	SUIT NO.	WICK	AIR VELOCITY (mph)	OUTER SLEEVE	T _s (°F)	T _{wick} (°F)	T _{os} (°F)	T _{air} (°F)	DEW POINT (°F)	%RH	COOLING RATE (BTU/hr/ft)
90		dry			92.2	89.9	86.8	84.8	-----	-----	13
91					91.0	86.4	83.2	85.1	63.5	48	96
92			0.83	12-in.	91.4	87.2	85.2	84.9	72	65	73.5
93					91.2	87.6	85.5	85.0	75.5	72.5	63
94		wet			91.5	86.2	83.2	84.9	60	42.5	106
95	8				91.3	86.4	80.9	84.8	64.5	49.3	103
96					91.5	86.3	80.8	84.7	64	48.7	102
97					91.8	87.6	83.5	84.6	71.5	63	81
98			1.38	11-in.	91.1	87.1	83.6	85.3	74.5	71	67
99					91.0	87.8	83.9	85.0	79	82	56
100					91.1	86.4	81.6	84.8	62	45	108

T_s: Simulated skin temperature (membrane)T_{wick}: Outer wick temperatureT_{os}: Outside surface temperatureT_{air}: Air temperature

*Calculated from wet-dry bulb measurements

**Natural convection

***Forced convection

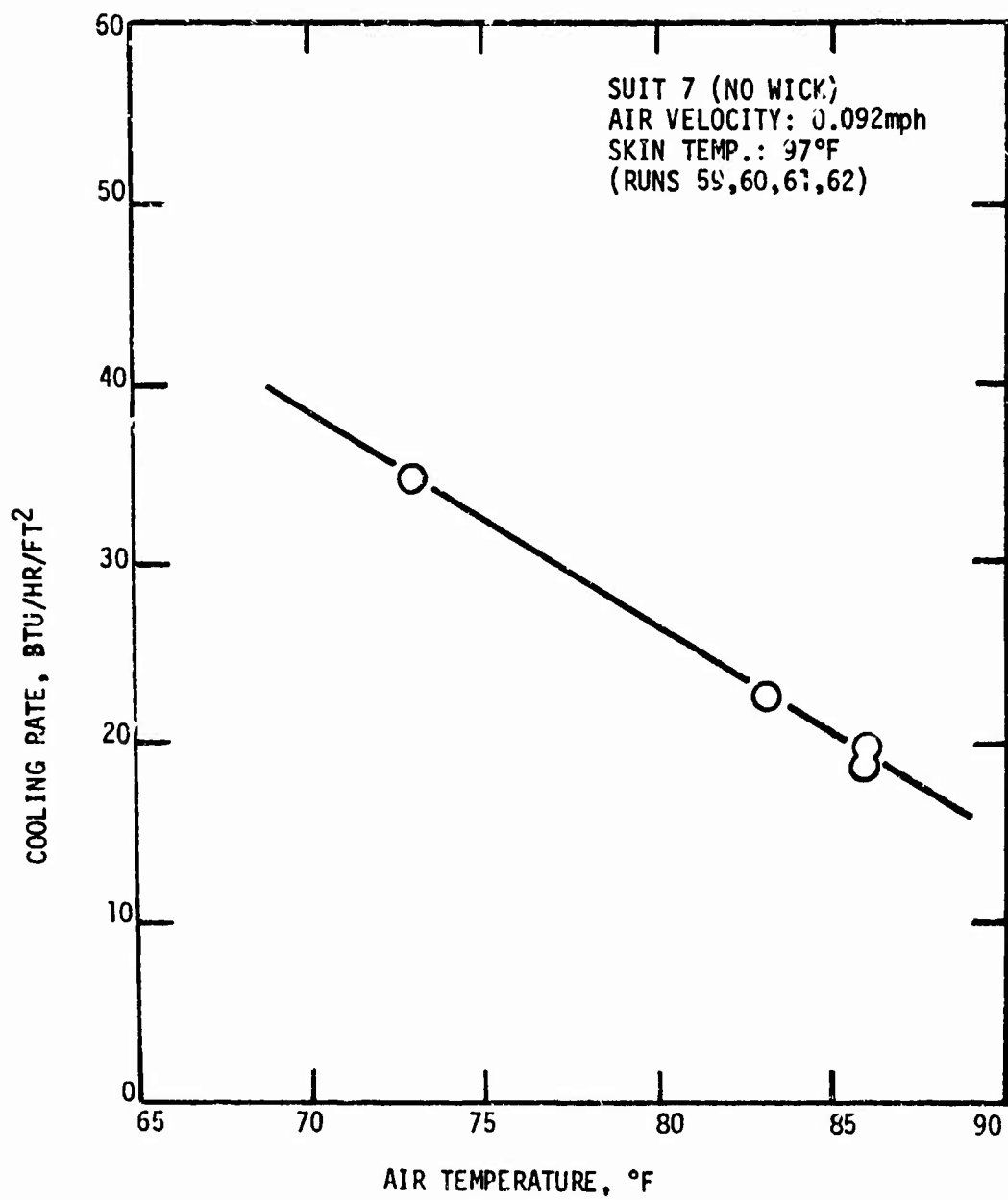


Figure 2. Effect of Air Temperature on Cooling Rate

The effect of air velocity is shown in Figure 4. Here the cooling rate (for air of 50 percent relative humidity) is plotted against the air velocity.

The results shown in Figure 3 are replotted in Figure 5 as cooling rate versus the water partial pressure driving force. The question originally arose as to where the bulk of the water vaporization was taking place. Since in many runs the surface temperature of the clothing was at a lower temperature than either the wick or the air, it was originally assumed that the bulk of the water was vaporizing on the surface. However, in calculating the partial pressure difference driving force, it was found that a better correlation resulted when it was calculated between the air and the wick rather than between the air and the outer surface of the clothing. The cooling rate versus the driving force is shown in Figure 5.

6. Summary

Equipment for testing the heat transfer characteristics of various clothing systems under a wide range of simulated climatic conditions, including the use of wet wicks to simulate sweating, was designed, constructed and successfully tested.

A carbon-treated, polyurethane nylon Tricot laminate cloth was tested for heat and water vapor transfer using the sweating cylinder under a wide range of conditions. Reasonably high cooling rates were obtained. With a skin temperature of 91°F and air at 85°F and 50 percent relative humidity flowing over the surface at 3.6 miles per hour, a cooling rate of 129 Btu per hour per square feet was obtained.

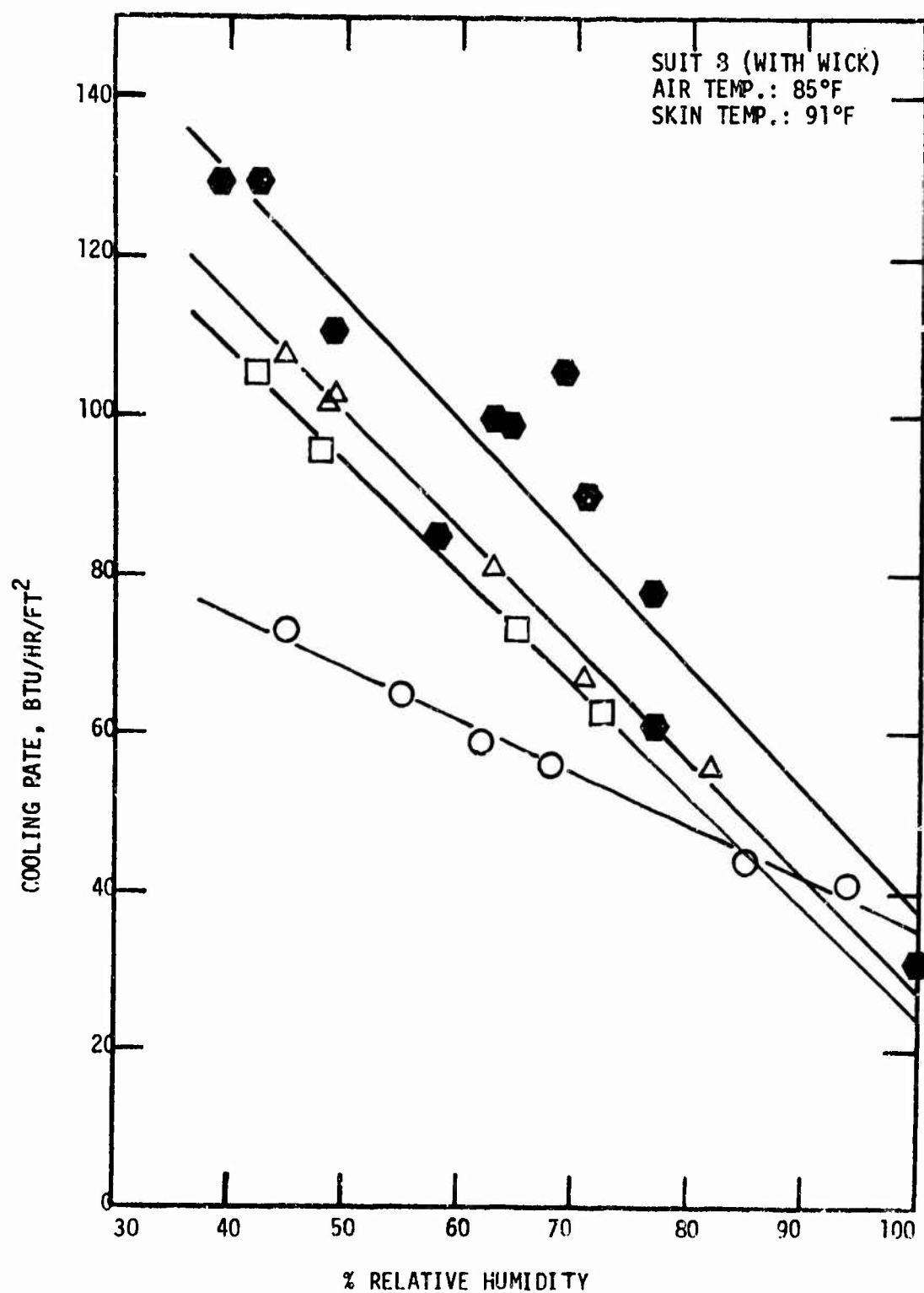


Figure 3. Effect of Air Relative Humidity and Air Velocity on Cooling Rate

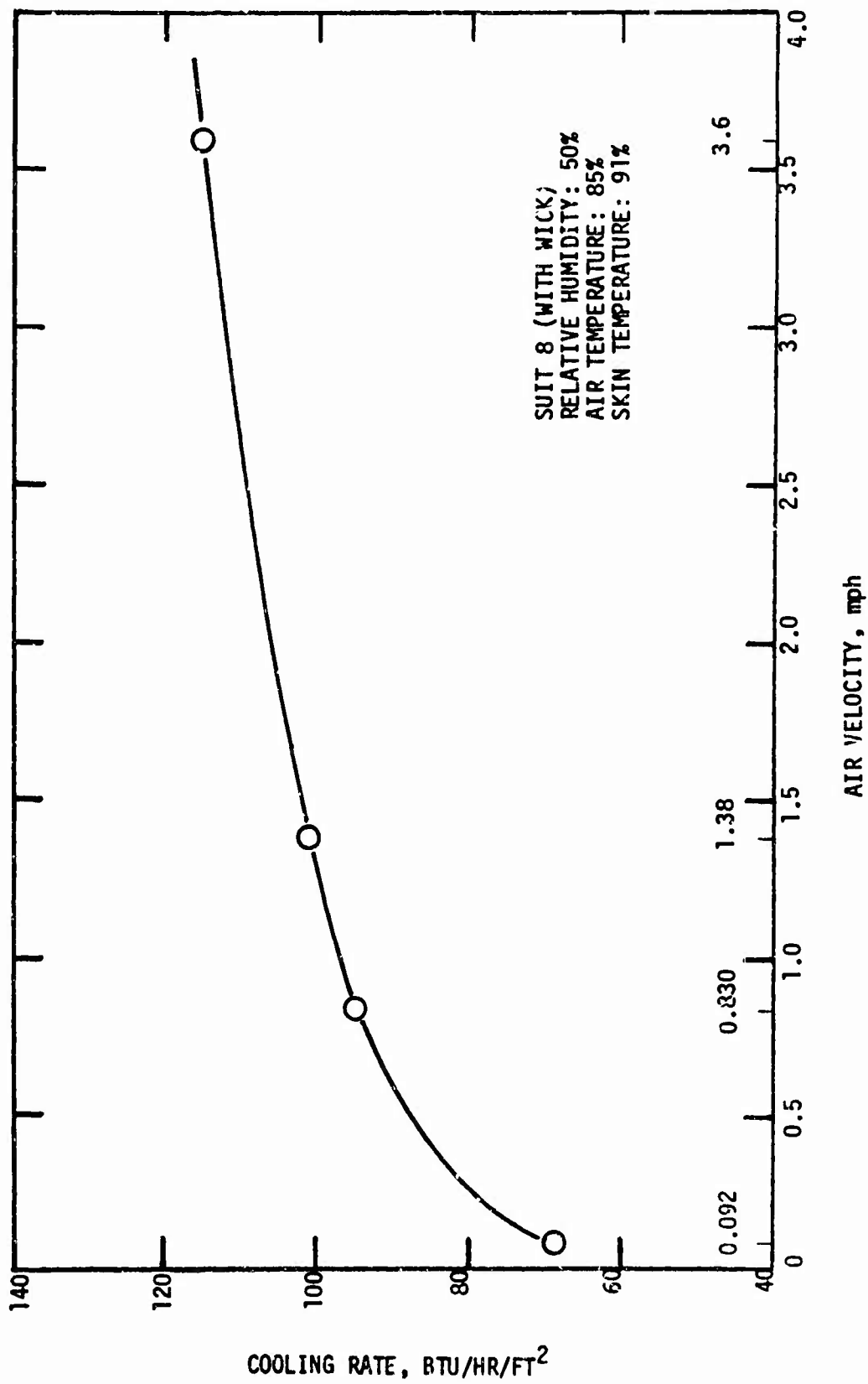


Figure 4. Effect of Air Velocity on Cooling Rate

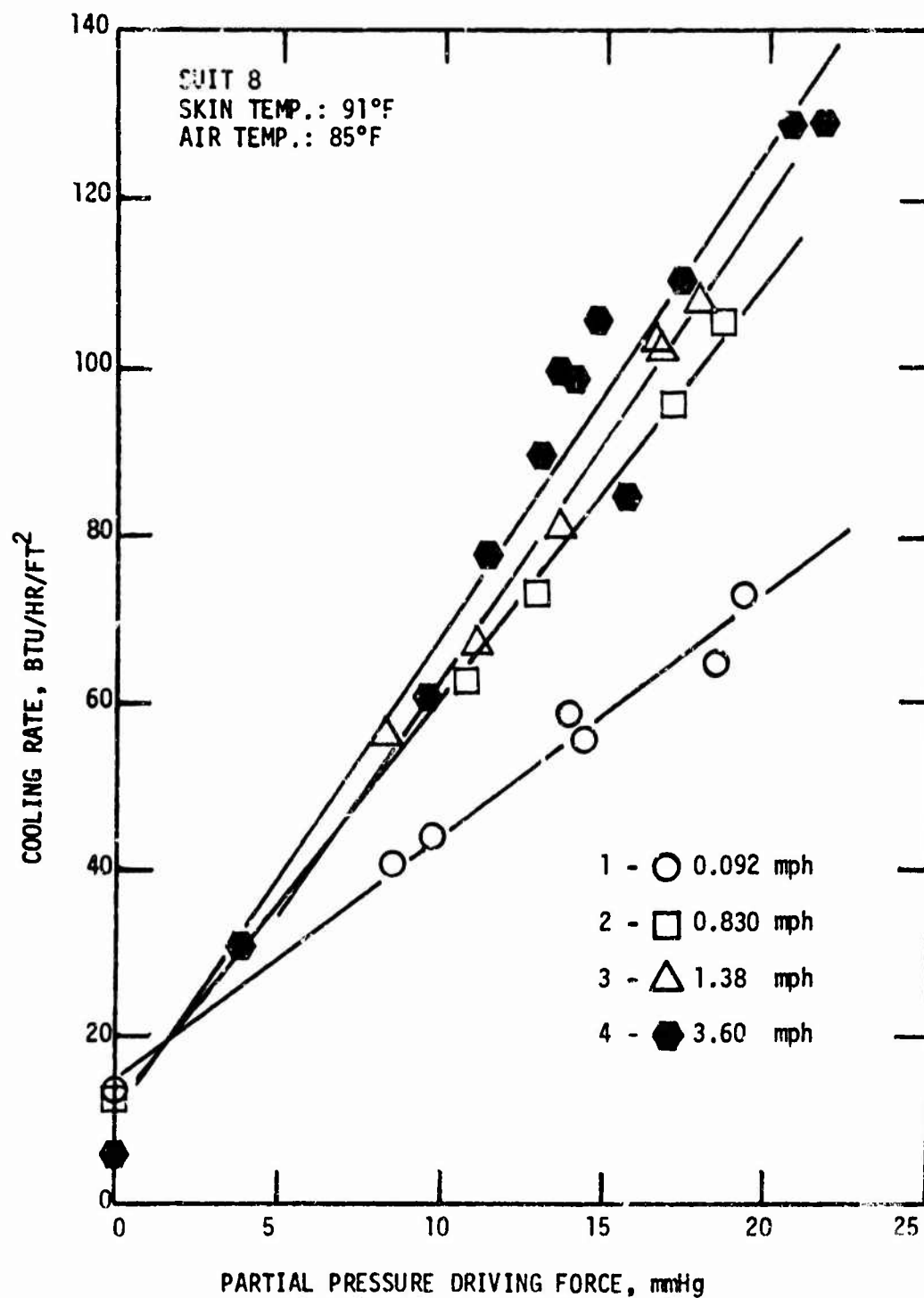


Figure 5. Effect of Water Partial Pressure Driving Force ($P_{wick} - P_{air}$) on Cooling Rate

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